

Experimental study of a thermoacoustically-driven traveling wave thermoacoustic refrigerator

B. Yu^{a,b}, E.C. Luo^{a,*}, S.F. Li^{a,b}, W. Dai^a, Z.H. Wu^a

^a Key Laboratory of Cryogenics, Technical Institute of Physics and Chemistry, Chinese Academy of Sciences, Beijing 100190, China

^b Graduate School of Chinese Academy of Sciences, Beijing 100049, China

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ABSTRACT

New configurations of traveling wave thermoacoustic refrigerators driven by a traveling wave thermoacoustic engine were introduced and tested in this paper. First, the performance of the refrigerator with different-diameter inertance tubes was investigated experimentally. Then, investigation of substituting a flexible membrane attached to inertial mass for inertance tube was tested. The experimental results show that the substitution could improve the efficiency of the system and lead to a larger cooling power. So far, using helium gas as the working gas, the system could provide 340 W cooling power at the temperature of $-20\text{ }^{\circ}\text{C}$ with working frequency of 57 Hz and average pressure of 3.0 MPa. The total COP, i.e. cooling power divided by heating power, is 0.16.

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1. Introduction

Thermoacoustic machines, which mainly include thermoacoustic heat engines and refrigerators, make use of thermoacoustic effect to realize the conversion between heat and sound energy. Before 1999, most of the studies focused on standing-wave thermoacoustic systems which have relatively simple structure but low thermal efficiency due to the inherent irreversibility of standing-wave conversion. Until 1999, Backhaus et al. developed a traveling wave thermoacoustic engine (TWTE) with a thermal efficiency of about 30% [1]. In the same year, Swift et al. studied a pulse tube refrigerator with acoustic power recovery of lost power in the orifice-type pulse tube cryocoolers [2], which is now called thermoacoustic-Stirling refrigerator or traveling wave thermoacoustic refrigerator (TWTR). In principle, the TWTR cannot only be applicable to achieve cryogenic temperature range below 120 K but also should be more efficient than those traditional pulse tube refrigerators including orifice-type, double-inlet type and inertance-type of pulse tube cryocoolers. Theoretically, at the refrigeration temperature of 100 K, the traditional pulse tube refrigerator have an ideal COP of only $1/3$ (i.e., $\text{COP} = T_c/T_h$) (assuming the ambient temperature of 300 K); however, the TWTR can achieve an ideal COP of $1/2$ (i.e., $T_c/(T_h - T_c)$) because it can recover the acoustical power dissipated in the hot-end components of the traditional pulse tube cryocoolers. In particular, for room-temperature cooling the TWTR can achieve much higher efficiency than the traditional pulse tube refrigerators because there is more

fraction of acoustical power dissipated pulse tube refrigerators [3]. In recent years, much effort has been done in developing TWTRs for higher cooling temperature range [4,5], in which the loudspeakers or linear compressors were used as the drivers.

As the TWTE and TWTR have promising potential in achieving higher efficiency, coupling the two can make a heat-driven and efficient refrigerating system with no moving components, which may find important applications in not only cryogenic cooling but also room-temperature cooling. In 2004, Yazaki et al. studied a system which has two regenerators in one loop to realize the functions of both the TWTE and TWTR [6], but the efficiency is very low because the only loop has no two locations to keep the pressure and velocity waves being in phase simultaneously. In 2006, a thermoacoustically-driven refrigerator with two separate traveling wave loops was built in our laboratory, and it obtained 250 W cooling power at the temperature of $-22.1\text{ }^{\circ}\text{C}$ and the total COP (i.e., the cooling power divided by the heating power) is about 0.12 [7]. The double thermoacoustic-Stirling system showed a much better cooling performance than any previously reported thermoacoustic refrigerators. However, there are still several possible improvements for the whole system: (i) a relatively long inertance tube used in the TWTR, which is also called inertance tube used for adjusting the phase relationship of pressure and velocity waves inside the regenerator and for recovering the acoustical power, may have two drawbacks: large flowing loss and incompactness. (ii) According to our separate test on the output characteristics of the TWTE [8,9], it can deliver a net acoustic power of more than 600 W. If the TWTR has a COP of more than 1 at the cooling temperature of $20\text{ }^{\circ}\text{C}$, its cooling power should go up to 600 W more or less. This means that the old TWTR did not match the TWTE. Thus, optimization of the

* Corresponding author.

E-mail address: Ecluo@mail.ipc.ac.cn (E.C. Luo).

impedance match between TWTE and TWTR may further improve the total COP and cooling power.

Thus, for building a compact TWTR and for improving its cooling performance, new configurations of traveling wave thermoacoustic refrigerators were introduced and tested in this paper. Firstly, a large TWTR was built, and it still used inertance tube as the phase shifter and the acoustical power recover. Secondly, a flexible membrane attached a mass is used to substitute for the inertance tube. The purpose of the substitution is to decrease larger flowing loss by the inertance tube. In the experiment, a significant improvement on the cooling performance was observed, and the cooling power and the COP were significantly improved compared with the inertance-type TWTR reported in this work and our previously reported TWTR.

The following section describes the details of the experimental setup. The third section gives the experimental results and discussions. Finally, some conclusions are made.

2. Experimental system

Fig. 1 shows the schematic of the traveling wave thermoacoustic refrigerator driven by a traveling wave thermoacoustic engine. The details of the TWTE can be found in Refs. [8,9], and extensive tests on the TWTE were conducted separately. For quick reference, some main structure parameters are still presented in the paper, shown in Table 1. The TWTR mainly includes a regenerator, ambient and cold heat exchangers, a thermal buffer tube, a compliance cavity and an inertance tube. The details of each part dimension are listed in Table 2.

In reference to Fig. 1, the first unique feature of this refrigerator is that the thermal buffer tube is actually composed of ten parallel tubes arranged around the system axis symmetrically. The hemispherical stuff shown at the bottom of the TWTR is used to reduce the dead volume between the thermal buffer tube and the cold heat exchanger. The second unique feature is that inertance tube is imbedded into the compliance cavity for the sake of compactness. Besides, a flexible membrane, which is put in the cavity near the ambient heat exchanger, is used to suppress Gedeon streaming [10].

According to Table 2, the ratio of the length to the diameter of this regenerator is only 0.4. So, multi-dimensional effects should be very obvious and may lead to a radial in-homogeneous temperature distribution inside the regenerator, which can result in loss of cooling power. For this reason, two pieces of copper screen are sandwiched between every ten pieces of stainless steel screen to increase radial heat conduction when packing a regenerator [11].

Table 1

Main structure parameters of the traveling wave thermoacoustic engine.

Main ambient heat exchanger	80 mm i.d., 48 mm long, parallel-plate type heat exchanger
Regenerator	80 mm i.d., 80 mm long, 120 mesh stainless steel screens
High temperature heat exchanger	80 mm i.d., 50 mm long, parallel-plate type heat exchanger
Thermal buffer tube	80 mm i.d., 240 mm long
Secondary ambient heat exchanger	80 mm i.d., 22 mm long, parallel-plate type heat exchanger
Inertance tube	80 mm i.d., 1700 mm long

Table 2

Details of the traveling wave thermoacoustic refrigerator.

Cold end heat exchanger	100 mm i.d., 30 mm long with about 0.315 m ² heat exchange area
Regenerator	100 mm i.d., length 40 mm, 120 mesh stainless steel screens. There are two pieces of 120 mesh copper screens between every ten pieces of screens
Ambient heat exchanger	100 mm i.d., 35 mm long with about 0.365 m ² heat exchange area
Thermal buffer tube	Ten 29-mm-diameter tubes arranged in parallel, length 410 mm
Compliance cavity	150 mm i.d., length 220 mm
Inertance tube	29–40 mm i.d., length 120 mm

In the experiment, an average pressure of 3.0 MPa helium gas is used. The dynamic pressure is measured by two pressure sensors P1 and P2 which are placed at the inlets of the refrigerator and the ambient heat exchanger in the engine loop, respectively. The heater block temperature of the engine is measured by a K-type thermocouple whose uncertainty is 1 K. Electrical heaters are placed at the cold heat exchanger to simulate cooling power. Two calibrated Pt sensors are symmetrically mounted on the cold heat exchanger to measure the cold end temperature. The accuracy of the Pt sensors is 0.1 K after calibration. Cooling water temperature of ambient heat exchangers is kept at around 15 °C.

3. Experimental results

3.1. Influence of the dimension of the inertance tube

In the TWTR, the inertance tube has two main functions: one is to serve as a passage for feeding back acoustic power and the other is to adjust the phase relationship between oscillating pressure and

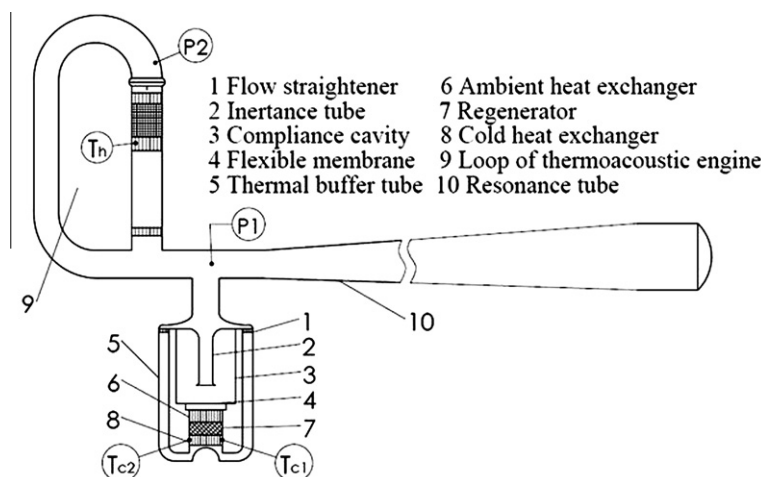


Fig. 1. Schematic of the traveling wave thermoacoustic refrigerator driven by the traveling wave thermoacoustic engine.

volume flow in the regenerator. Radebaugh noted that the loss in the regenerator is least when the phase relationship between the pressure and volume is in phase with each other near the center of the regenerator [12]. For this purpose, length and diameter of the inertance tube should be carefully selected. In our experimental setup, the length of the inertance tube is restricted by the length of the compliance cavity. With the same length of 120 mm, we mainly investigated how the inertance tube diameter (varying from 29 mm to 40 mm) influences the system performance.

Fig. 2a–c give the total COP (i.e., cooling power divided by heating power) at different cold end temperature with different inertance tubes and heating powers. The maximum heating power, varies from 2000 W to 2600 W, is limited due to the material limitation of the heater block, where the temperature is always controlled around 660 °C. From the figure, it can be seen that the refrigerator with about 35-mm-diameter inertance tube leads to the best performance, and the COP decreases gradually with the increase of heating power. According to Fig. 2c, the maximum COP at

0 °C, –20 °C, –40 °C are 0.154, 0.106, 0.075 respectively. The corresponding cooling powers are 366 W, 260 W, 173 W. This performance is not satisfying and the total COP is a bit lower than that of the refrigerator in Ref. [7]. Moreover, Fig. 3 gives the pressure ratio at the inlet of the refrigerator when the cooling temperature is –20 °C. Obviously, the pressure ratio increase with the increase of heating power and diameter of inertance tube.

To investigate possible reasons for the unsatisfactory results, a numerical simulation based on thermoacoustic theory for the refrigerator is conducted according to Ref. [13]. It is found that the flow in the inertance tube already entered the regime of turbulence. Under such circumstance, the acoustic power loss inside the inertance tube might be serious and was estimated by the simplified turbulent flow model according to Ref. [14]. In addition, the minor loss in the interface between the inertance tube and the compliance cavity can be evaluated by the following equation [15].

$$\Delta \dot{E}_2 = \frac{2}{3\pi} \frac{K \rho_m}{A^2} |\tilde{U}|^3$$

where, K is the dimensionless minor loss coefficient; ρ_m is the static density of the gas and $\tilde{U}$ is the volume flow rate. Thus, the total loss of the inertance tube can be estimated with measured pressure ratio as shown in Fig. 3. The calculated results are shown in Fig. 4, which indicate the loss of the inertance tube is significant. Even more than one fifth of the input power of the TWTR was dissipated in the inertance tube, and the loss increase with more heating power and smaller diameter of inertance tube. It should be noted that, the total loss mainly caused by the entry and exit effects.

In addition, because of larger resistance of the turbulent flow, the real phase relationship in the refrigerator loop may not be as good as predicted. So, we decided to use inertial mass to provide inertance for the loop to see how it influences the system performance.

3.2. Substitution with the inertial mass for the inertance tube

Fig. 5 shows the photo of the inertial mass. It consists of two thin metal pieces sandwiching a piece of flexible membrane by small bolts, which is actually a mass-spring combination. Fig. 6 is the schematic of the refrigerator with an inertial mass. The assembly of the inertial mass and the membrane not only can provide necessary inertance for the TWTR but also can suppress the 2Gedeon stream in the traveling wave loop. Consequently, the original membrane placed nearby the ambient heat exchanger can be removed. In the experiment, the inertial mass was placed about 2 cm from the inlet of the cavity to ensure the volume of the cavity is same as that when using inertance tube.

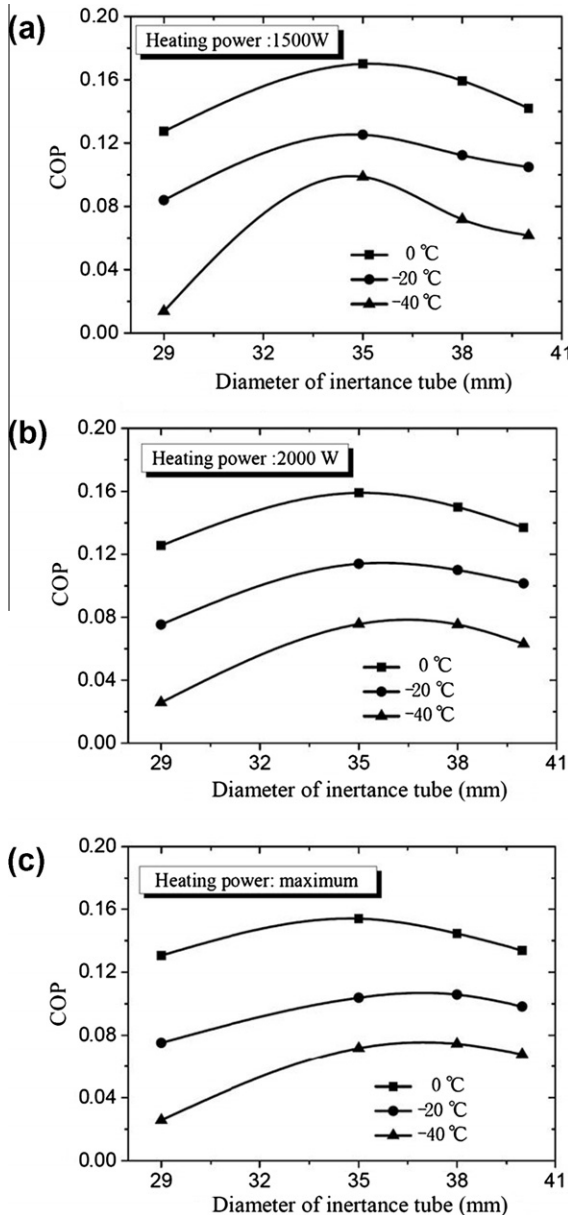


Fig. 2. The total COP with different heating powers: (a) 1500 W, (b) 2000 W and (c) maximum heating power to keep the heater block at 660 °C.

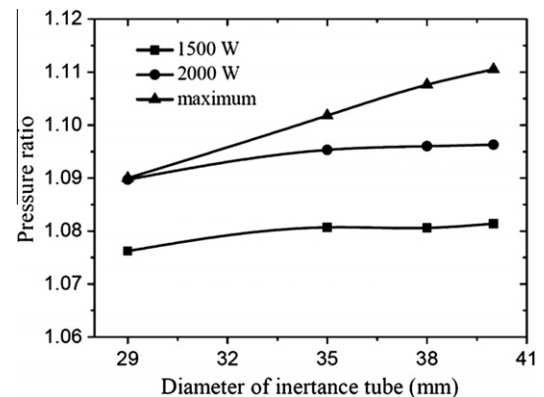


Fig. 3. The inlet pressure ratio of the TWTR with different inertance tube and heating power when the cold end temperature is –20 °C.

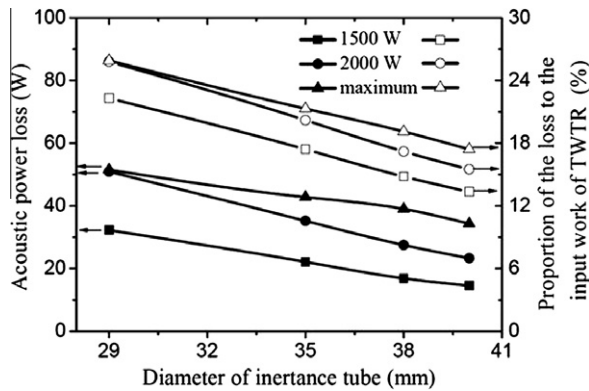


Fig. 4. Calculated acoustic power loss of the inertance tube with the inlet pressure ratio as shown in Fig. 3.



Fig. 5. The photo of the mass adhered to the membrane as an inertial element.

It is important to choose a proper weight of the inertial mass for efficient operation. Fig. 7a and b shows the experimental results with different inertial mass weights and heating powers. The maximum heating power varies from 1750 W to 2400 W, which is reduced by 200 W compared to that with inertance tube. From the figure, it can be seen the cooling performance improve significantly. According to Fig. 7b, the maximum COP at 0 °C, –20 °C, –40 °C are 0.216, 0.16, 0.112, respectively. In other words, the COP increased even more than 50% compared to the refrigerator with an inertance tube. The corresponding maximum cooling powers are 469 W, 340 W, 230 W, respectively, almost 30% improvement. The cooling performance is also much better than our previously reported results in Ref. [7] under the same TWTE.

There must be other reason for the huge improvement because the inertance tube consumed only about 20% of the inlet acoustic power of the refrigerator. In fact, in order to provide the same inertance as the inertial mass, the diameter of the inertance tube should be small enough (less than 35 mm) which will bring serious acoustic work loss at the same time. So, a larger diameter of the inertance tube would be better. It means the optimal inertance of the inertance tube is less than that of the inertial mass, thereby influencing the load of the TWTR. Notably, the pressure ratios shown in Fig. 8 are generally less than that in Fig. 3, which also indicate the load of the TWTR has changed. The change of load

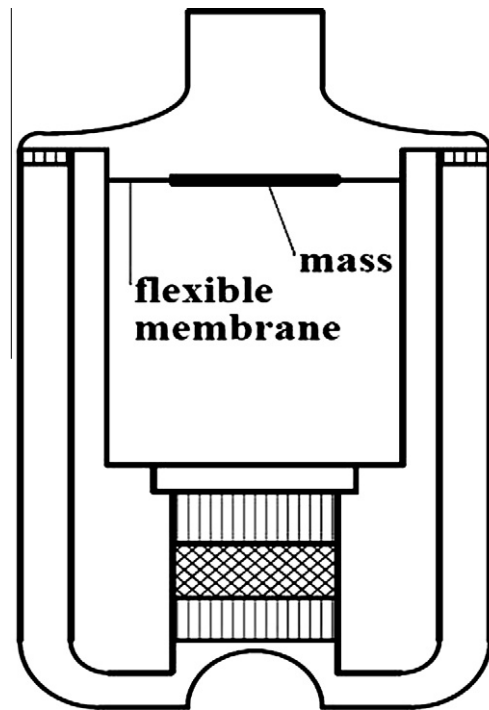


Fig. 6. Schematic of the refrigerator with an inertial mass instead of the inertance tube.

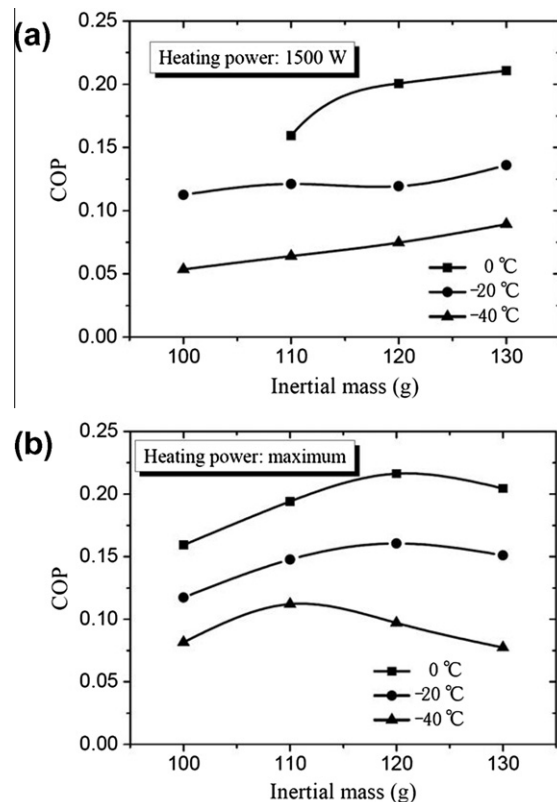


Fig. 7. Cooling power curves with different heating power: (a) 1500 W and (b) maximum heating power to keep the heater block at 660 °C.

would influence the output efficiency of the engine (i.e., output acoustic power divided by heating power).

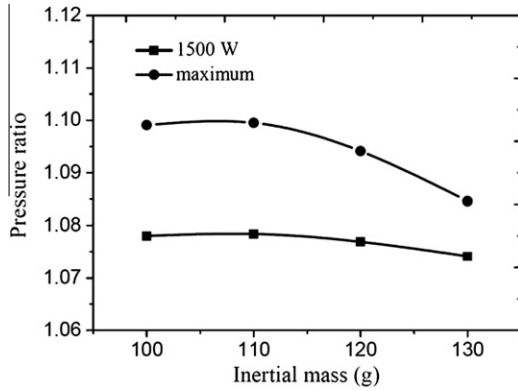


Fig. 8. The inlet pressure ratio of the TWTR with different inertial mass and heating power when the cold end temperature is -20°C .

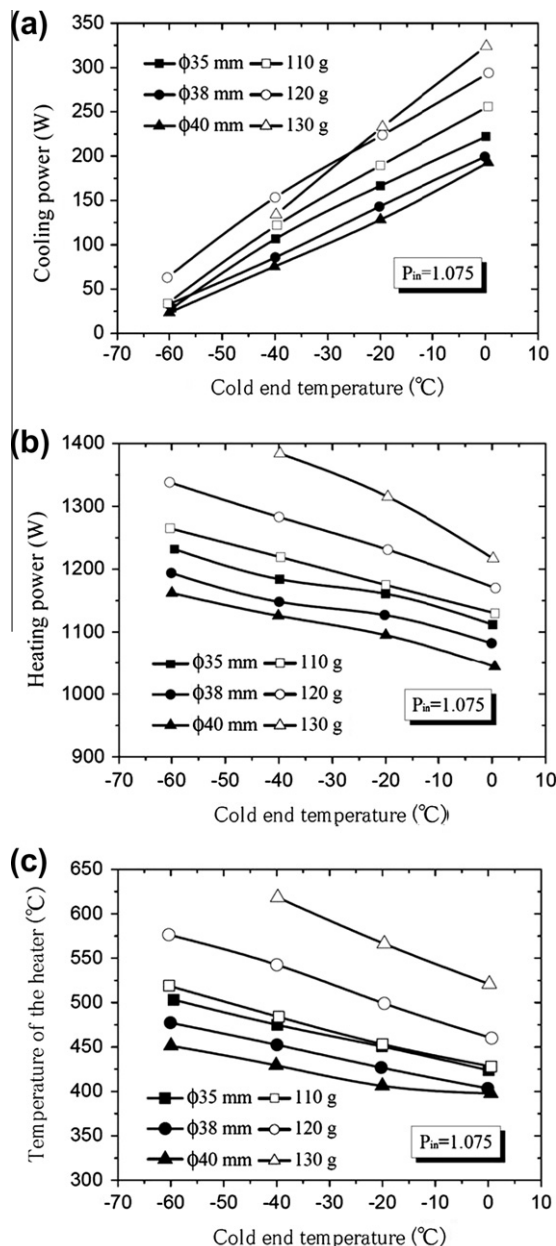


Fig. 9. With the pressure ratio of 1.075 at the entrance of the refrigerator, compare: (a) cooling power, (b) heating power and (c) temperature of the heater block.

The experimental comparisons of the refrigerator with two kinds of inertial elements at the same inlet pressure ratio of 1.075 can give a better understanding of the improvement. The results are shown in Fig. 9a–c. According to Fig. 9a, the maximal cooling powers obtained by the refrigerator with inertial mass are generally about 1.4 times larger than that with inertance tube. This does not mean the COP of the refrigerator subsystem also improves so much since we only fix the refrigerator inlet pressure ratio, not the input acoustic power. Actually, at the same cold end temperature and inlet pressure ratio, the system using inertial mass requires more heating power as shown in Fig. 9b and leads to a higher temperature of the heater block as shown in Fig. 9c. It implies more acoustic power was generated by the engine and the increased acoustic power was mainly consumed by the refrigerator. It also means the output efficiency of the engine improved.

In a word, the substitution of inertial mass for the inertance tube not only decreases the acoustic work loss, increasing the COP of the refrigerator, but also improves the output efficiency of engine. Both of the two factors lead to the huge improvement above mentioned.

4. Conclusion

A new configuration of compact traveling wave thermoacoustic refrigerator driven by a traveling wave thermoacoustic engine has been built and many experiments have been done to test the performance. The two phase shifters with inertance tube and lumped inertial mass were tested and compared experimentally. It is found that the TWTR with the assembly of lumped inertial mass and flexible membrane as the phase shifter achieved an obvious improvement in cooling performance; in addition, the experiment showed that the compact system worked reliably. With an inertial mass adhered to a piece of flexible membrane replacing the inertance tube, the maximum cooling powers of 340 W and 469 W are obtained when the cold end temperature are -20°C and 0°C , respectively, and the corresponding total COPs are 0.16 and 0.216. This cooling performance is among the best experimental results reported so far, and it shows more possibility toward practical applications of the TWTR.

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